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# Experimental Testing of Prototype Face Gears for Helicopter Transmissions

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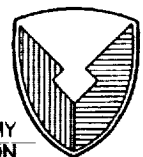
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(NASA-TM-105434) EXPERIMENTAL  
TESTING OF PROTOTYPE FACE GEARS FOR  
HELICOPTER TRANSMISSIONS (NASA)

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US ARMY  
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# EXPERIMENTAL TESTING OF PROTOTYPE FACE GEARS FOR HELICOPTER TRANSMISSIONS

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## ABSTRACT

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An experimental program to test the feasibility of using face gears in a high-speed and high-power environment was conducted. Four face gear sets were tested, two sets at a time, in a closed-loop test stand at pinion rotational speeds to 19,100 rpm and to 271 kW (364 hp). The test gear sets were one-half scale of the helicopter design gear set. Testing the gears at one-eighth power, the test gear set had slightly increased bending and compressive stresses when compared to the full scale design. The tests were performed in the NASA Lewis spiral bevel gear test facility. All four sets of gears successfully ran at 100 percent of design torque and speed for 30 million pinion cycles, and two sets successfully ran at 200 percent of torque for an additional 30 million pinion cycles. The results, although limited, demonstrated the feasibility of using face gears for high-speed, high-load applications.

## INTRODUCTION

Current helicopter drive system designs are striving to further reduce weight to improve helicopter performance and capabilities. In the recent studies that have recently been conducted (ref. 1) and those currently ongoing (refs. 2 to 4), improved designs have shown that torque splitting can have a dramatic weight savings. A current candidate to achieve torque splitting has been through use of face gears. In this design, a single spur gear pinion splits the load to two output face gears.

The face gear design has been discussed by several investigators (refs. 5 to 7); however, their use has typically been for low power applications. Only recently has this gear design been considered for possible high-speed and high-power applications. One problem with using this design is that design data or accepted design practice is nonexistent. Since their application in the past has been limited to low power and speed, only dimensional information is available from handbooks such as reference 8. Recent work has centered on the

design (ref. 3) and on describing the surface geometry (ref. 9). The surface geometry, based on the manufacturing process, is necessary to predict the tooth contact of the face gears. From the tooth contact analysis made via reference 9, adjustments to the machine tool settings are made until the desired contact locations are determined. This procedure was used to design the face gears used in this experimental program.

The objectives of this work are to describe the preliminary experimental results of a face gear design in a high-speed, high-load environment. This work was conducted as part of a joint effort between the U.S. Army Propulsion Directorate, NASA Lewis Research Center, McDonnell Douglas Helicopter Co., Lucas Western, Inc., and the University of Illinois at Chicago in support of the U.S. Army's Advanced Rotorcraft Transmission Program (ART). Face gears were tested in the NASA Lewis spiral bevel gear rig. Basic face gear design, testing procedures, test facility, and results of test are presented. Recommendations for face gear improvements are made.

## APPARATUS

### Test Hardware

Face gears can be utilized when a drive system must transmit power between intersecting shafts. The pinion member is a simple spur gear. The gear member can be thought of as a rack gear with connecting ends (fig. 1). Across the gear member tooth profile from the inner to outer radius, the pressure angle and tooth normal section vary continuously. If the outside diameter is to the maximum allowable during manufacture, the teeth will be pointed.

The face gear design parameters used in this study are shown in table I. The test hardware is shown in figure 2. The gears were basically a one-half scale version of the face gear stage in the ART split-torque transmission proposed by the McDonnell Douglas and

Lucas Western team (ref. 3). The reduction ratio of the face gear design was 3.82:1. Since the gears were designed to approximately the same pinion bending stress levels as the full-scale version, the maximum design power of the test set was about one-eighth that of the full-scale design. A 90° shaft angle was used to facilitate installation and operation in the existing Lewis spiral bevel test rig (described later). Maraging 300 steel was chosen because of its low heat treat distortion characteristics. The pinion was made using standard aerospace practices where the surfaces were nitrided and ground. The measured surface hardness of the pinion was  $R_c$  58. The gear member was cut via a shaper and hardened. No final grinding of the gear profile was done since current manufacturing methods do not exist to grind face gears. The measured surface hardness for the gear was  $R_c$  52.

### Design Considerations

The selection of a design using face gears was driven by the advantages which would accrue if the concept proved successful. The concept selected has split-torque load paths, which provide significant weight savings. In the ART concept (ref. 3), the input spur pinion is free to float radially. It is located radially by balancing the opposing mesh forces of two face gears simultaneously engaged, which provides the split-torque load paths. The input spur pinion is also free to float axially, which is a characteristic of face gears. Thus, the input spur pinion and its shaft are light and simple. Face gears have true conjugate action. The ART design chosen has a high contact ratio analogous to helical gearing. Thus, low vibration and noise are expected. However, several areas required investigation. As noted, the analytical basis for design was not sufficient for aerospace gearing. The torque capacity of the ART face gear design required test substantiation. The mode of failure was unknown.

Analytical support was provided by Litvin, et al. (ref. 9). The analytical work included determining outer and inner radii of face gear teeth, tooth contact analysis, contact ratio, and the effects of misalignment and eccentricity, and evolving a process for grinding face gear teeth.

The design investigations indicated that face gears had an improved weight advantage compared to spiral bevel gears at ratios higher than approximately 3.5:1. A simplified rule-of-thumb was developed that allowable face width multiplied by the diametral pitch should be greater than seven, with 10 indicating clear superiority

to spiral bevel gears. The ART design uses 1.25 in. face width at 8 pitch and a reduction ratio of 3.82:1, thus meeting both criteria. The face gear test reported herein preserved this geometry, using a face gear width of 0.62 in., diametral pitch of 16, and a reduction ratio of 3.82:1. The face gear teeth were slightly crowned using a conventional face gear manufacturing technique. While the input spur pinion has 28 teeth, the shaper cutter had 29 teeth. When a 29 tooth shaper cutter generated the face gear teeth, a slight crowning resulted. The crowning was intended to keep the contact pattern in the middle of the teeth. Misalignment of the gear axes in the absence of crowning could cause the contact pattern to concentrate at the edges of the teeth, resulting in early failure. The crowning does result in an increased contact stress because the contact area is reduced compared to a properly aligned uncrowned tooth. Nevertheless, crowning was selected to avoid the hazard of misalignment.

The test torque was selected to exceed the ART design stresses (108 percent of bending stress, 103 percent of contact stress). The anticipated mode of failure was bending failure of the pinion teeth, clearly geometrically weaker than the face gear teeth. To this end the test torques and durations were selected to test for infinite bending fatigue life ( $30 \times 10^6$  cycles) at design bending stress, followed by doubling test torque and running to  $30 \times 10^6$  cycles or failure.

### Test Facility

The test facility used for this face gear study is the NASA Lewis spiral bevel gear test rig. An overall sketch of the rig is shown in figure 3(a). In this facility two gear meshes, in this case face gears, are tested simultaneously. One side operates in the typical fashion of the pinion driving the gear (test section), and on the other side (slave side) the gear drives the pinion. The rig operates in a closed loop fashion whereby the drive motor supplies the desired shaft speed and the facility power losses. Loop power is controlled by a thrust piston that moves a helical gear axially. The loop load is read out directly via a torque meter that is part of the loop shafting. The facility is also equipped with a speed sensor, thermocouples, flow meters, pressure transducers, and accelerometers (fig. 4).

Facility operational parameters during the face gear tests are shown in table II. Lubricant type used during the test was DOD-L-85734. Four jets, 0.762 mm (0.03 in.) diameter, lubricated the pinion and gear at into-and out-of-mesh locations.

## Face Gear Setup and Contact Pattern

The face gears were set up in the test facility just as spiral bevel gears. The only difference is that axial positioning of the spur pinion only would effect where on the pinion, the contact would take place. Moving the gear member axially was the only adjustment mechanism. This was used to set the backlash to 0.051 to 0.102 mm (0.002 to 0.004 in.). A red lead compound also was used to evaluate the composite of the tooth surface contacts. A no-load contact pattern attained for build 1 slave gear side is shown in figure 5. All other contact patterns were similar to the one shown.

## Test Procedure

Due to the uncertain capabilities of the face gear system at high speed and load, a gradual breakin procedure was adopted on the first build of the facility and the procedure was as follows: The gears were run at a low speed and torque condition for about 10 min. The facility was then shut down, and the gears were visually inspected. The speed and/or torque was then increased, and the procedure was repeated for all break-in conditions in table III. After the breakin procedure, the tests were conducted on the two builds as shown in tables III and IV. Load and speed conditions were maintained within  $\pm 3$  percent of the nominal settings during all the tests conducted.

The anticipated failure mode of the face gear set was tooth breakage of the pinion member. The generally accepted knee of the S-N endurance curve is 30 million cycles. Tests were conducted at 100 and 200 percent load in an attempt to meet this endurance requirement.

## RESULTS

### 100 Percent Torque Performance Tests

After accumulation of 30 million pinion cycles, the test hardware was removed and photographed. Both builds with the test hardware successfully completed this phase of the testing. The test and slave section hardware of the second facility build are shown in figures 6 and 7 and figures 8 and 9, respectively. After 30 million pinion cycles, both test and slave side pinions for both builds showed only minimal surface change from their as-manufactured condition. The face gear condition was different from test to slave section side. For both builds, the test side was generally in good condition, with small areas of micropitting randomly scattered over the active profile. The slave side had significantly more

surface distress. On the slave side, for the first build, a couple of teeth had pit lines across the face width of the teeth centered between the root and tooth tips. Most teeth had micropitting near the root area of the active profile. For the second build, the slave side had pit lines across the face width on most of the teeth. The cause of the difference in appearance between the test and slave side face gears are not known at this time.

### 200 Percent Torque Performance Tests

After successful completion of the first series of tests at 100 percent torque, the torque was increased in 20-percent increments until the 200-percent torque level was reached. The gears were visually inspected after 1 hr of operation at each increased level of torque for the first build (table III). The torque was increased from 100 to 200 percent load in one step for the second build (table IV). The test hardware from the first build is shown after 30 million pinion cycles at 200 percent torque in figures 10 to 13.

In the first build all hardware survived the 200 percent torque test. Slightly more wear on all members was apparent from that of the tests conducted at 100 percent torque. The pinions and the test side face gear had moderate wear and some random micropitting, but generally were in good shape for a 200-percent torque test. The slave side face gear surface distress, as previously mentioned (pit lines and micropitting), became more pronounced. However, there was no damage that would preclude further testing.

Since the first build tests showed that the design could tolerate operation at 200 percent torque, the second build hardware torque was increased from 100 to 200 percent torque in one step. After operation at this load, the facility was stopped at approximately 12 million pinion cycles, and a tooth failure of the slave gear was found. This is shown in figure 14. The failure appeared to start at the pitting line near the tooth top and propagated through the tooth. Only a slight vibration increase was noted during this testing period. The failure was found by a visual inspection after a normal shut down. The other components of the second build looked basically the same as after the 100 percent torque tests.

### Vibration Results

Two piezoelectric accelerometers were mounted on the rig. One at each location near the test and slave side pinion support bearings (fig. 4). The accelerometers measured vibration in the vertical direction and had a resonant frequency of 27 kHz. The accelerometer

outputs were fed to charge amplifiers which had band-pass filters (from  $\approx 100$  to 5000 Hz). Typical vibration spectra are shown in figure 15. The data shown were taken during the first build of the facility. Even with the bandpass filtering, the signals had much activity outside the 100- to 5000-Hz range. The vibration levels at the face gear meshing frequency (8917 Hz) and first harmonic (17 833 Hz) were rather low. This was consistent with the testing as the rig ran rather quiet and smooth. The major component of the measured spectra is believed to be from the pinion ball bearing outer-race frequency (1622 Hz) and harmonics.

### Noise Results

During facility operation at various levels of speed and load, a sound power level (SPL) measurement was made with a hand-held instrument. The peak sound pressure level measurements are shown in table V. These measurements were taken during the first build of the facility as shown in table III. The measurement was taken with the instrument approximately 15 cm (6 in.) from the rig front face Lexan cover in alignment with the gear axis of rotation (see fig. 4). The results shown in table V indicate that the peak SPL was insensitive to load once the 100 percent torque level was reached.

A comparison of the face gear results to previously taken data on this facility with spiral bevel gears would not be appropriate. Since the designs differ so much, it would be better to run a test that would provide a fair comparison where the design application was fixed and a best design for both gear types then could be made.

### CONCLUSIONS

Overall, the face gears operated smoothly and quietly. Therefore, face gears have demonstrated the ability to operate at high speed and high torque. The pinions and the test side face gears showed some wear and micropitting after the 200-percent torque tests, but were generally in good shape. However, the slave side face gears showed considerable surface distress. It is not known why the slave side face gears exhibited more surface distress than the test side face gears. In operation in the NASA test rig the test side operates with the spur gear pinion driving the face gear, and on the slave side the face gear drives the spur pinion. In the facility both sides have the tooth contact occurring on the same sides of the teeth; however, the operation is reversed. This might be one cause for the discrepancies between the two sides. Another possibility might be the rig itself,

with one side being more susceptible to wear than the other. There was, however, no indication of this from previous test performed on spiral bevel gears in the same rig. A third cause of discrepancies might just be the randomness of endurance testing.

As previously mentioned, the face gears were not ground since the manufacturing capability does not exist to grind face gears. If such a technique did exist, the face gears could be hardened and ground to produce an extremely hard and smooth gear, comparable to conventional aerospace quality gearing. This would significantly improve the surface durability limit and possibly eliminate the surface relate distress found in the present tests.

### SUMMARY OF RESULTS

Experimental tests were performed on face gears to demonstrate operation in a high-speed, high-load application. The gears were a one-half scale version of those designed for a helicopter main rotor transmission. The gears were tested in the Lewis spiral bevel gear rig. Endurance test were performed on four sets of gears. The following results were obtained:

1. Feasibility of face gears for high-speed, high-load applications such as helicopter transmissions was demonstrated. All four sets of gears were successfully run at 100 percent design torque for 30 million pinion cycles. Two of the four sets were successfully run at 200-percent torque for 30 million pinion cycles. One face gear had a broken tooth after about 12 million pinion cycles at 200-percent torque.

2. The failure mechanism of the face gears in these tests was tooth surface fatigue. The no-load contact pattern on the teeth was centered on the tooth surface, and contact occurred under load over the full active profile of the face gear. The pinion teeth showed normal wear. The face gear teeth, however, had some surface distress. The teeth from the test side pinion driving the the face gear, had moderate wear and were in good condition. The teeth from the slave side face gear driving the pinion, had small pit lines in some instances in the middle region of the teeth. In one test the slave side gear pitting lead to tooth failure.

3. The face gears ran relatively quietly and smoothly. The vibration at the face gear fundamental and harmonic frequencies were low compared to the overall test rig vibration.

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TABLE I.—FACE GEAR DESIGN DATA

AGMA quality	12
Number of teeth pinion, gear	28,107
Diametral pitch	16
Pressure angle, deg.	25.0
Shaft angle, deg.	90.0
Face width, mm (in.), of —	
Pinion	37.6 (1.285)
Gear	15.5 (0.62)
Hardness of pinion gear, $R_c$	58, 52
RMS surface finish, $\mu\text{m}$ ( $\mu\text{-in.}$ )	0.51 (20)
AGMA pinion bending stress index, MPa (ksi)	248 (36)
AGMA pinion contact stress index, MPa (ksi)	1034 (150)
Gear material	Maraging 300 steel*

\*Per AMS 6514.

TABLE II.—TEST FACILITY  
OPERATIONAL PARAMETERS

Pinion shaft speed (100%) rpm	19,110
Pinion torque (100%), N·m (in·lb)	67.8 (600)
Nominal lubricant flow rate cm/s (gpm)/mesh	51 (0.8)
Lubricant oil inlet temperatures, °C (°F)	74 (165)
Lubricant jet pressure, MPa (psi)	0.552 (80)

TABLE III.—TEST OPERATING CONDITIONS BUILD 1

Run	Time		Face gear speed		Face gear torque		
	Million pinion cycles	hr	rpm	Percent of maximum	N·m	in·lb	Percent of maximum
1 (break-in)	0.3	0.28	1000	20	65.5	580	25
	.2	.17	1000	20	129.9	1150	50
	↓	.17	1740	35	65.5	580	25
		.17	1740	35	129.9	1150	50
		.18	2480	50	129.9	1150	50
		.20	2480	50	195.5	1730	75
		.17	2480	50	259.1	2293	100
	.3	.23	3730	75	259.1	2293	100
	.2	.17	5000	100	259.1	2293	100
2	30	26.2	5000	100	259.1	2293	100
3 (overload)	1.2	1.05	5000	100	309.5	2740	120
	↓	1.03	↓	↓	361.6	3200	140
		1.00			413.0	3655	160
		1.03			464.4	4110	180
	30	26.2	↓	↓	518.1	4586	200



TABLE IV.—TEST OPERATING CONDITIONS BUILD 2

Run	Time		Face gear speed		Face gear torque		
	Million pinion cycles	hr	rpm	Percent of maximum	N·m	in.·lb	Percent of maximum
1 (break-in)	0.2 ↓	0.17	1000	20	65.5	580	25
		.20	1000	20	115.2	1020	45
		.17	1760	35	65.5	580	25
		.17	1760	35	128.2	1135	50
2	31.2	27.2	5000	100	258.7	2290	100
3 (overload)	12.0	10.5	5000	100	518.6	4590	200

TABLE V.—SOUND PRESSURE LEVEL (SPL) MEASUREMENT  
PEAK  
[Facility lubricant pumps running.]

Pinion speed <sup>a</sup>	Pinion torque <sup>b</sup>	Test side SPL, dB	Slave side SPL, dB
0	0	88	88
50	50	96	96
100	100	113	114
↓	120	113	↓
	140	114	
	160	113	
	180	114	
	200	113	

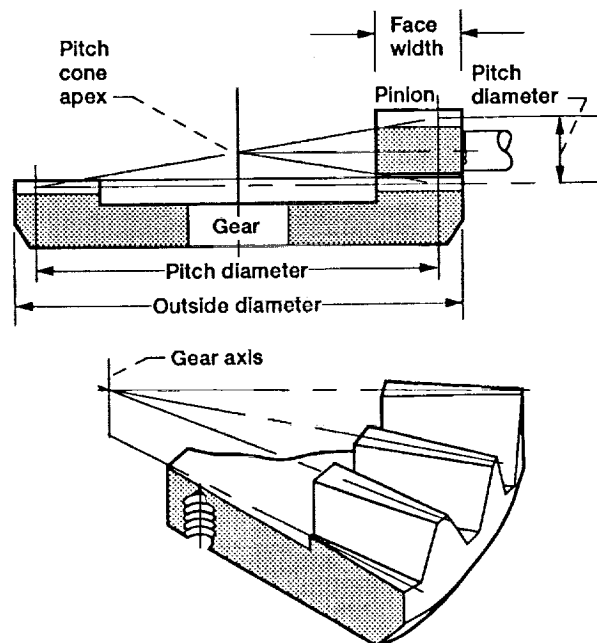
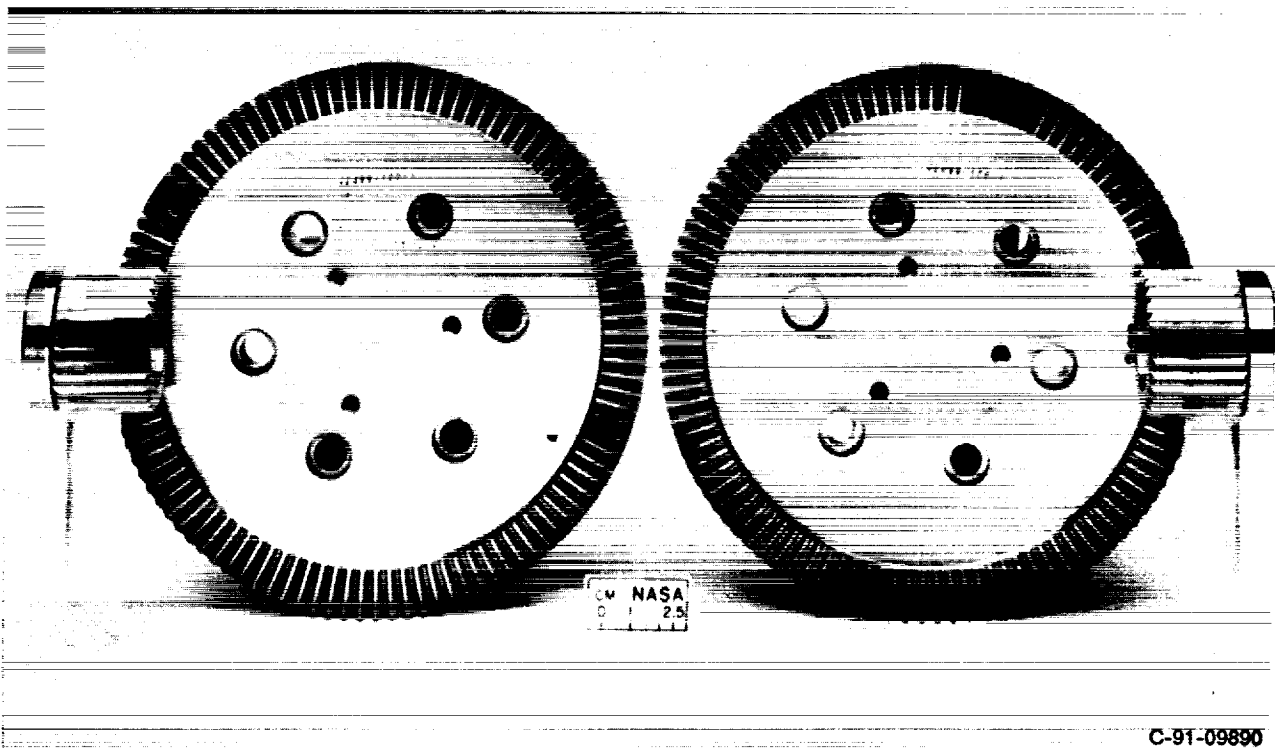
<sup>a</sup>Values are in percent of 19 110 rpm.<sup>b</sup>Values are in percent of 67.8 N·m (600 in.·lb).

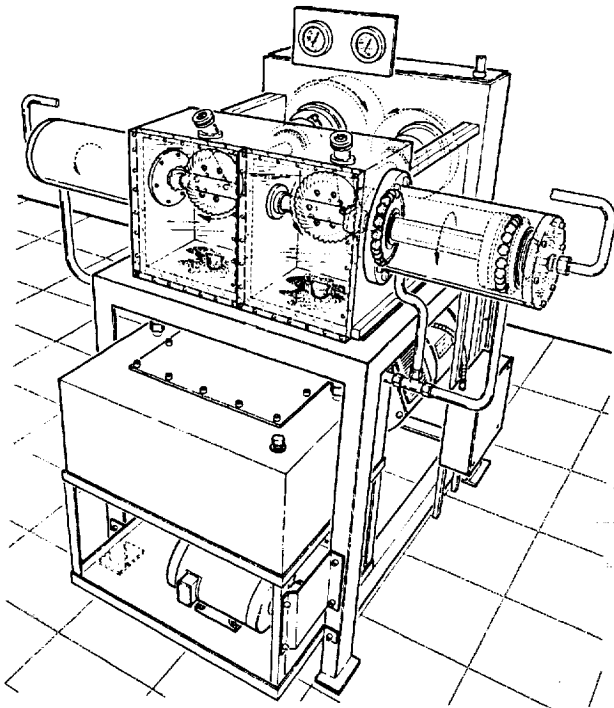
Figure 1.—Face gear terminology.

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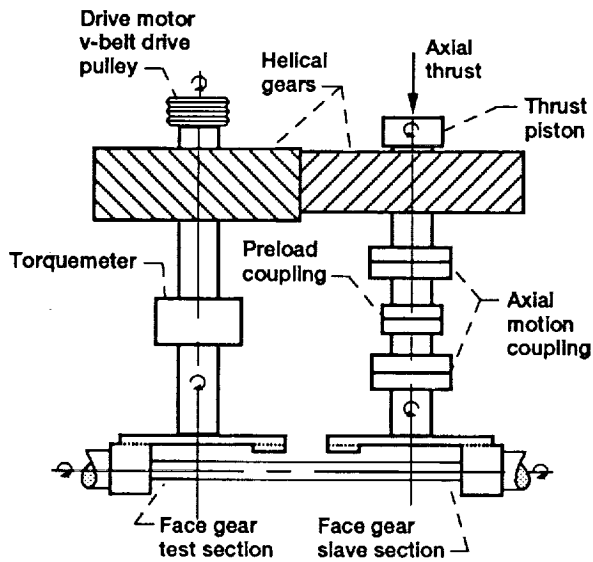


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Figure 2.—Face gears tested in the NASA Lewis spiral bevel gear rig.



(a) Spiral bevel gear configuration of facility.



(b) Facility layout in face gear configuration.

Figure 3.—Facility for face gear tests.

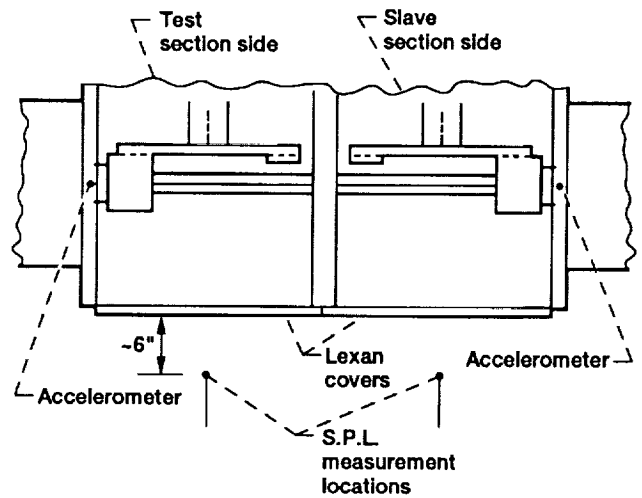


Figure 4.—Accelerometer locations and sound pressure level measurement locations (top view).

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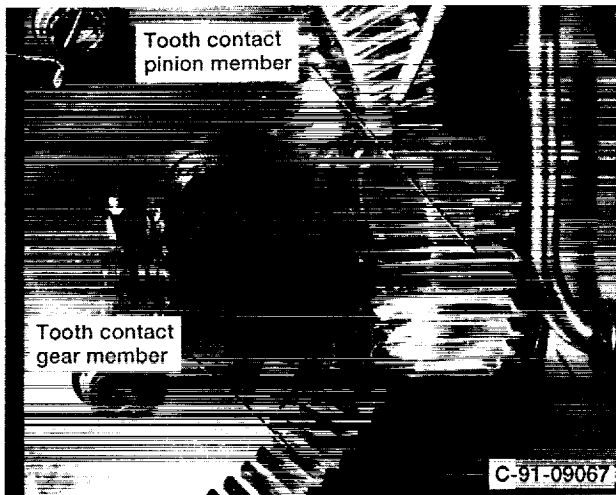


Figure 5.—Photograph showing contact pattern using red lead.

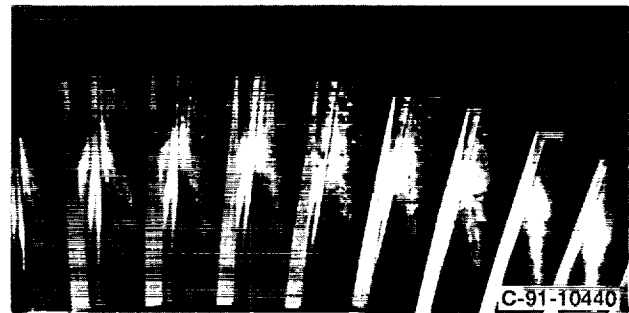


Figure 6.—Test side face gear after 30 million pinion cycles at 100 percent torque.



Figure 7.—Test side spur pinion after 30 million cycles at 100 percent torque.

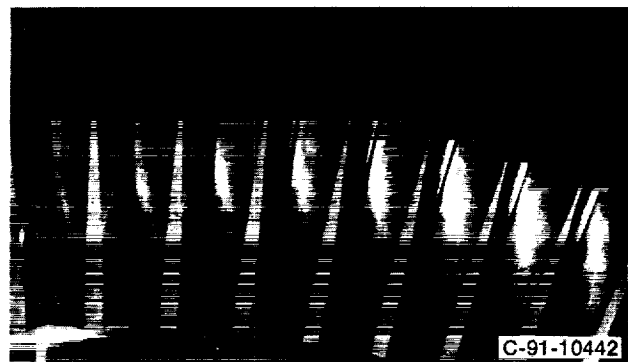


Figure 8.—Slave side face gear after 30 million pinion cycles at 100 percent torque.

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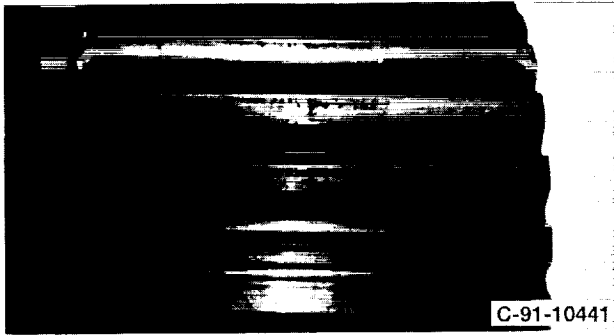


Figure 9.—Slave side spur pinion after 30 million cycles at 100 percent torque.



Figure 10.—Test side face gear after 30 million pinion cycles at 100 percent torque.

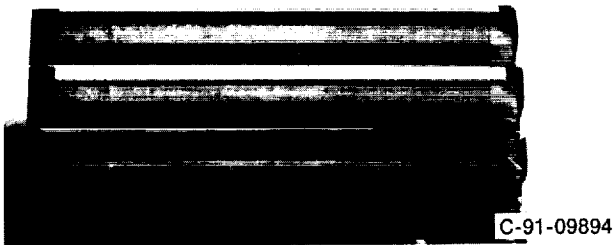


Figure 11.—Test side spur pinion after 30 million cycles at 200 percent torque.



Figure 12.—Slave side face gear after 30 million pinion cycles at 200 percent torque.

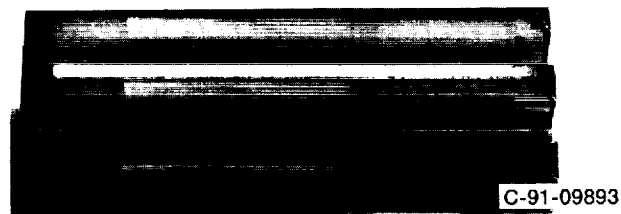
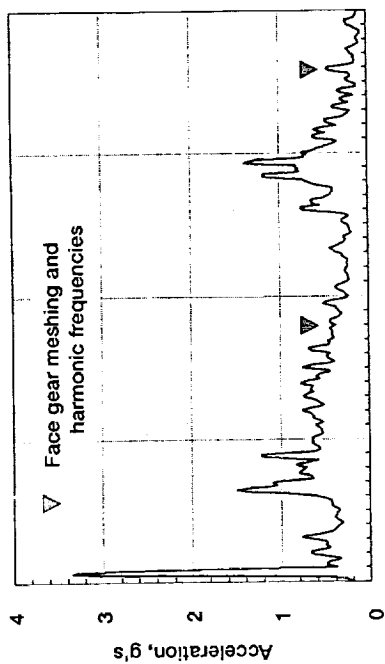


Figure 13.—Slave side spur pinion after 30 million cycles at 200 percent torque.

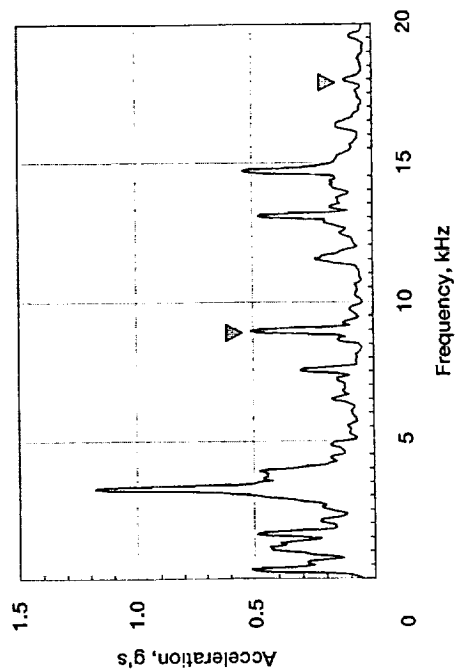
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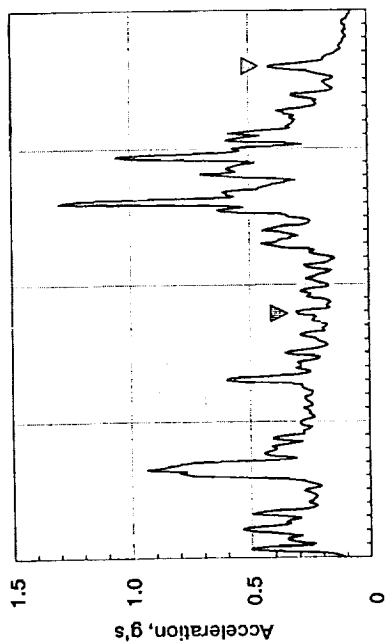
Figure 14.—Broken slave side face gear tooth after 8 million cycles at 200 percent torque.



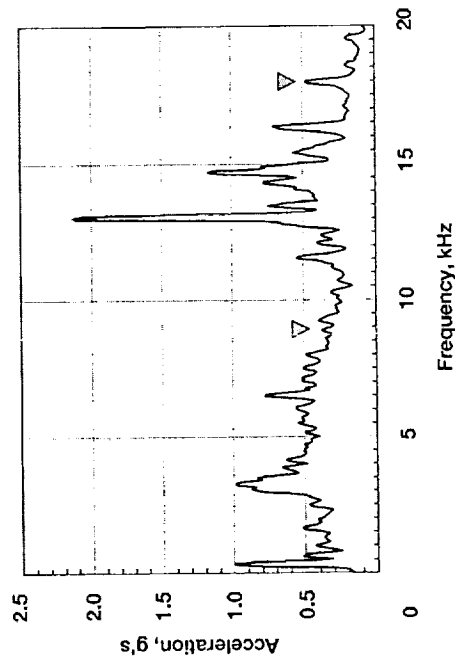
(b) Slave side accelerometer, 100% torque.



(d) Slave side accelerometer, 200% torque.



(a) Test side accelerometer, 100% torque.



(c) Test side accelerometer, 200% torque.

Figure 15.—Vibration spectrum of face gear tests.

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13. ABSTRACT (Maximum 200 words)  An experimental program to test the feasibility of using face gears in a high-speed and high-power environment was conducted. Four face gear sets were tested, two sets at a time, in a closed-loop test stand at pinion rotational speeds to 19 100 rpm and to 271 kW (364 hp). The test gear sets were one-half scale of the helicopter design gear set. Testing the gears at one-eighth power, the test gear set had slightly increased bending and compressive stresses when compared to the full scale design. The tests were performed in the NASA Lewis spiral bevel gear test facility. All four sets of gears successfully ran at 100 percent of design torque and speed for 30 million pinion cycles, and two sets successfully ran at 200 percent of torque for an additional 30 million pinion cycles. The results, although limited, demonstrated the feasibility of using face gears for high-speed, high-load applications.				
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